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# ENHANCEMENT OF CO<sub>2</sub> REFRIGERATION CYCLE USING AN EJECTOR: 1D ANALYSIS

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## ABSTRACT

An ejector expansion transcritical CO<sub>2</sub> refrigeration cycle is proposed to improve the performance of the basic transcritical CO<sub>2</sub> cycle by reducing the expansion process losses. A 1D analysis is carried out in order to investigate the ejector performance using supercritical CO<sub>2</sub> as fluid. The entrained flow condition is analyzed assuming constant pressure mixing inside the constant area section of the ejector. A sensitivity study is performed to investigate the effect of inlet temperature and pressure on the cycle performance.

The COP of the ejector expansion transcritical CO<sub>2</sub> cycle can be improved by more than 15% compared to the conventional transcritical cycle for typical air conditioning operating conditions. A comparison between ejector refrigeration cycle using R-134a and CO<sub>2</sub> as refrigerants shows a better performance for CO<sub>2</sub> refrigerant.

## 1. INTRODUCTION

Throttling loss in the expansion device, through which the refrigerant is expanded from the condenser pressure to the evaporator pressure, is one of the thermodynamic losses in a conventional vapor compression refrigeration cycle. This expansion results during the isenthalpic process in which the kinetic energy developed as the refrigerant pressure decrease is dissipated to the refrigerant as friction heat. The isenthalpic process causes a larger amount of the refrigerant to flash into a vapor than in the isentropic process. As a result, the refrigerating effect of the cycle is reduced. In order to recover the potential kinetic energy in the expansion process, various researchers have attempted to use other expanders rather than the expansion engine (Disawas, S., and Wongwises, S., 2004; Nickl, J. et al., 2005; Rusly, E., et al, 2005).

Rusly, E., Aye, L., Charters, W.W.S., Ooi, A., 2005. Due to the low cost, no moving parts and ability to handle two-phase flow without damage, an ejector is an attractive alternative for the expansion device in the refrigeration system.

A transcritical CO<sub>2</sub> refrigeration cycle with constant pressure-mixing zone ejector (expansion device) is analyzed and simulated using a one-dimensional model. Different operating conditions are considered and the cycle performance with a CO<sub>2</sub> ejector is compared to the conventional CO<sub>2</sub> refrigeration cycle and a conventional R134a system. In addition, an ejector is designed for given operating conditions.

## 2. EJECTOR CYCLE

The ejector is a component that expands a high-pressure primary substance to absorb a secondary substance at a pressure slightly above the low pressure reached by the primary substance. In refrigeration cycles, the two substances are identical, so both flows mix together leading to mixture pressure increase due to the change of the flows momentum.

An ejector is composed of a nozzle and a body. The nozzle is convergent divergent with a throat that defines the primary mass flow rate. The role of the nozzle is to create a low-pressure flow with high momentum, so it transforms the pressure potential energy into kinetic energy.

The body of the ejector shape defines the ejector operation mode: "constant pressure mixing" ejector and "constant area mixing" ejector. The constant pressure ejector body is composed of a convergent to assure a constant pressure for the two flows before entering a constant area throat where the mixture of flows should occur at constant

pressure. However, in the constant area-mixing ejector, the two flows enter directly in a constant area region and the mixture of flows does not occur at constant pressure. After the constant area, a diffuser is installed for both types to decelerate the mixture flow and increase the ejector outlet pressure. The role of the body is to define the secondary inlet area, to assure the mixture between the flows and to transform the kinetic energy into pressure potential energy (Chunnanond, K., and Aphornratana, S., 2004a, and Chunnanond, K., and Aphornratana, S., 2004b).

The high-pressure fluid, known as "primary fluid", expands, accelerates through the nozzle and exits with supersonic speed to create a very low-pressure region at the nozzle exit plane and hence in the mixing chamber. Having a pressure difference between the expanded flow and the low-pressure side ejector inlet flow, known as "secondary fluid", the low-pressure fluid is entrained into the mixing chamber. The primary fluid expansion continues (in a fictive cone) without mixing with the secondary fluid. At some cross-section along this duct, the speed of secondary fluid may reach sonic velocity and chokes.

Experience shows that the constant pressure-mixing ejector offers better performances compared to the constant-area ejector (Huang, B.J., and Chang, J.M., 1999).

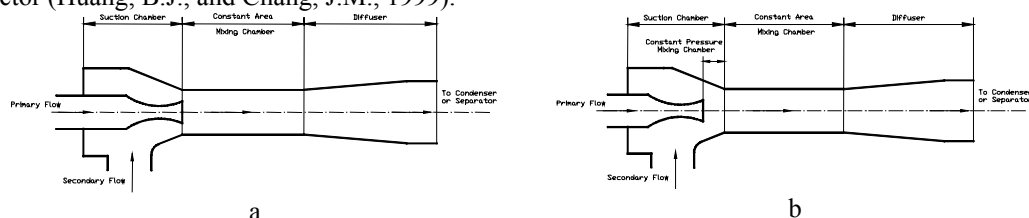


Figure 1: Schematic view of an ejector: a- constant area mixing ejector; b- constant pressure mixing ejector.

The ejector cycle has two operation modes: superheated vapor expansion, and sub-cooled expansion (Figure 2), which is studied in this paper.

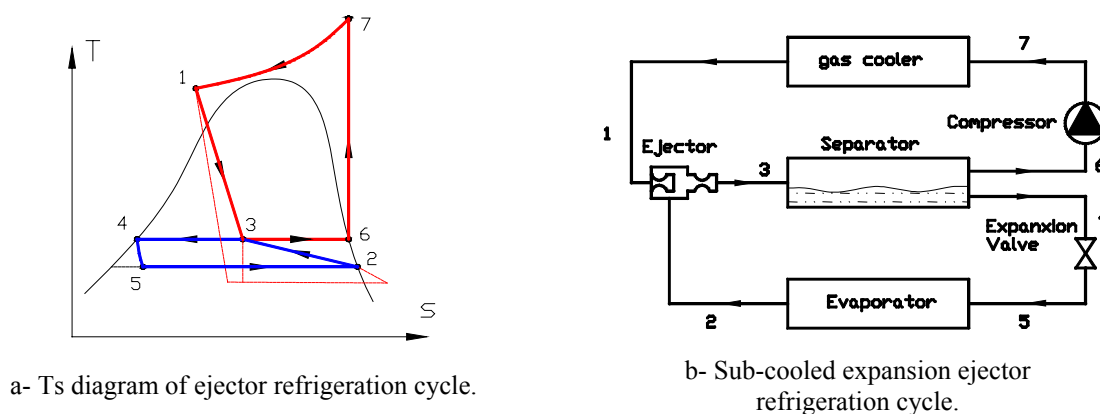


Figure 2: Schematic of ejector refrigeration cycle.

The sub-cooled (or supercritical) expansion ejector cycle (Fig. 2), is a two-phase expanding process; it is a promising cycle to enhance the performance of the conventional refrigeration cycle.

An ejector is analog to a turbo-machinery system, whereby a turbine mechanically drives a compressor through a common shaft. In the analog turbo-machinery (Figure 3), the high pressure fluid, the primary fluid, expands through the turbine to a pressure lower than the evaporator pressure, then the generated work is used to compress both fluids, primary and secondary, to an intermediate pressure.

In the ideal analog turbo-machinery of an ejector (Fig. 4), the primary fluid expands to the intermediate pressure through the turbine while the compressor compresses the secondary fluid to the intermediate pressure, the discharge from the compressor and the discharge from the turbine combine to form the discharge mixture (equivalent to the discharge from an equivalent ejector) (Li, D., and Groll, E. A., 2005).

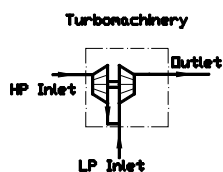


Figure 3: Ejector analog turbo compressor.

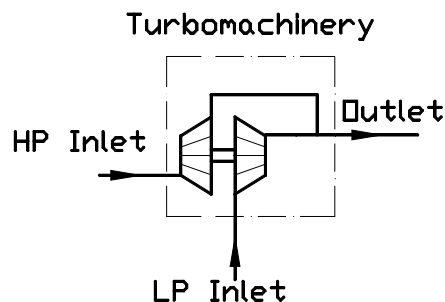


Figure 4: Ideal analog turbo-compressor.

### 3. 1D SIMULATION OF AN EJECTOR CYCLE WITH LIQUID / SUPERCRITICAL INLET AS PRIMARY FLUID

A model has been elaborated to study the behavior of a sub-cooled expansion ejector refrigeration cycle, using mass, momentum and energy conservation equations.

To simplify the model of the ejector expansion refrigeration cycle, the following assumptions are made:

1. Neglecting friction at the walls, the pressure drop in the gas cooler and evaporator and the connection tubes.
2. All the components are thermally insulated, so there are no heat losses to the environment from the system except the heat rejection in the gas cooler. The ejector is adiabatic, rigid and impermeable.
3. The vapor stream from the separator is saturated vapor and the liquid stream from the separator is saturated liquid.
4. The flow across the expansion valve is isenthalpic.
5. The compressor has a constant isentropic efficiency independent of the compression ratio or the compressor speed.
6. The evaporator outlet is one phase: either saturated vapor or superheated vapor, the gas cooler outlet temperature is determined by the ambient temperature.
7. The flow in the ejector is a one-dimensional homogeneous equilibrium flow and steady throughout the ejector. All fluid properties are uniform across their respective cross-sectional area.
8. The primary stream and the secondary stream reach the same pressure at the inlet of the constant area mixing section of the ejector. There is no mixing between the two streams before the inlet of the constant area mixing section. The inlet velocities of the primary and secondary flows are negligible. Homogeneous equilibrium flow conditions are considered at the nozzle outlet in the primary flow.
9. The isentropic expansion efficiencies of the primary stream and secondary stream are constant. The ejector diffuser has a constant isentropic efficiency.

Considering the above assumptions and applying the conservation equations on the different control volumes of the ejector shown in Figure 5, the ejector refrigeration cycle equations have been developed.

The pressure in the control volume CV a-a'-b is considered constant and equal to  $P_L$ . The pressure  $P_L$  is lower than the evaporator pressure  $P_2$ , the pressure drop  $\Delta P$  is the driving potential of the secondary stream flow. The entrainment ratio  $w$  is given as the mass ratio between the primary and secondary streams.

The gas cooler exit temperature  $T_1$  and pressure  $P_1$ , the evaporation temperature  $T_s$ , the superheat at the evaporator outlet  $TS$ , the nozzle efficiency  $\eta_n$ , the secondary stream expansion efficiency  $\eta_s$ , the diffuser efficiency  $\eta_d$  and the compression isentropic efficiency are input values.

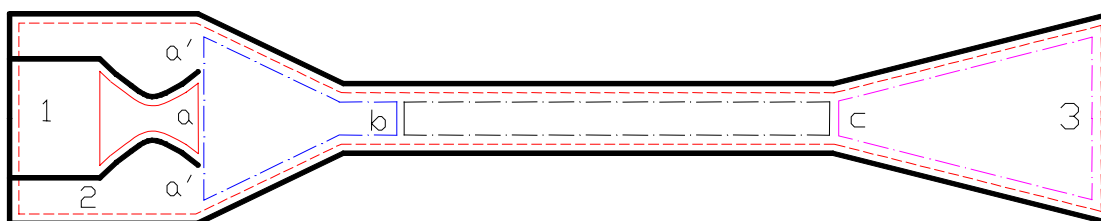


Figure 5: Specification of control volume for the one-dimensional flow model.

For the parameters mentioned previously, the COP of the ejector expansion cycle depends on the high pressure and the pressure drop  $\Delta P$  between  $P_{ev}$  and  $P_L$ .

The compression ratio of the ejector is defined as:

$$CR = P_3/P_2 \quad (1)$$

For  $CO_2$ , the conventional COP depends on the gas cooler pressure. An optimum pressure exists allowing maximum COP for a specified evaporation temperature, gas cooler exit temperature and isentropic compression efficiency. So, the comparison factor  $F$  between the conventional transcritical cycle and the ejector one is considered in the case of optimum pressure.

$$F = \frac{COP_{ejector}}{COP_{conventional}} \quad (2)$$

Considering the above theoretical model, the influence of pressure drop  $\Delta P$  and gas cooler pressure on the comparison factor is investigated using REFPROP 7 to calculate the refrigerant thermodynamic properties. The study is extended to evaluate the influence of different operating conditions for a given pressure drop  $\Delta P$  on the comparison factor.

#### 4. RESULTS AND DISCUSSION

The following standard operating conditions are assumed for calculations:  $T_{gc,out} = T_1 = 35^\circ C$ ,  $T_{ev} = T_5 = 2^\circ C$ ,  $TS = 5$  K. The ejector efficiencies are:  $\eta_n = \eta_s = 0.85$ ,  $\eta_d = 0.75$ . The isentropic compression efficiency is assumed equal to 0.8.

For these operating conditions, Table 1 compares the performance and the optimum operating pressure of gas cooler for different refrigeration cycles. The isentropic expansion efficiency is taken equal to 0.7.

Table 1: Comparison of different  $CO_2$  transcritical cycle performances

Refrigeration cycle configuration	COP	COP with isentropic expansion without work recovery	$P_{optimum}$ MPa	$P_{intermediate}$ MPa
Conventional $\eta_{is, comp} = 0.8$	2.820		8.7552	
Conventional with isentropic expansion	3.728	2.986	8.4658	
Two stages of compression in series with inter-cooling	4.498		8.7032	7.300
Two stages of compression in series with isentropic expansion and inter-cooling	7.117	4.606	8.2428	7.078
Two stages of compression with injection	3.249		8.4898	5.552
Two stages of compression with injection and isentropic expansion	4.21	3.354	8.3225	5.623
The ejector transcritical refrigeration cycle $\Delta P = 0.3467$ MPa, $w = 0.5353$ , $P_{mix} = 3.917$ MPa.	3.696		8.513	4.4

The consequence of the pressure drop  $\Delta P$  in the receiving section of the ejector  $P_{ev} - P_L$  on the  $F$  factor is shown in Fig. 6. It can be seen that for the given conditions, the ejector expansion transcritical  $CO_2$  cycle can achieve more than 30 % COP improvements over the conventional transcritical  $CO_2$  cycle for an optimum pressure drop of 347 kPa that depends on the nozzle geometric properties and the operating conditions of the ejector. Increasing the pressure drop increases the ejector compression ratio and decreases the compressor compression ratio and improves the cycle performances but shows an optimum CR value at 1.198. The compression ratio achieves the optimum value at the optimum COP, then, the performances tend to decrease while the pressure drop increases. The optimum gas cooler pressure varies from 8.65 to 8.51 MPa (delta lower than 0.14 MPa) and the entrainment ratio  $w$  varies from 0.54 to 0.532, (delta less than 0.8) for the studied variation. The ejector outlet quality is around  $0.65 \pm 0.0025$ .

The influence of the gas cooler pressure on the comparison factor  $F$  and the compression ratio  $CR$  of the ejector transcritical  $CO_2$  cycle are shown in Fig. 7. The two cycles are very dependent on the gas cooler pressure, and show an optimum gas cooler pressure at optimum performances. It can be seen that the comparison factor  $F$  of the ejector

expansion transcritical CO<sub>2</sub> cycle decreases with the increase of the gas cooler pressure. At the optimum pressure, factor F is not at its maximum value; the optimum enhancement varies from -4 % to 50% according to the pressure drop. As the gas cooler pressure increases, the compression ratio CR decreases to a minimum value that corresponds to the optimum gas cooler pressure, and then continues to slightly increase with pressure for different pressure drops. Factor F and compression ratio CR increase with the pressure drop to reach the optimum pressure drop (that is above 0.1 MPa). The optimum pressure and COP of the ejector cycle are respectively, 8.643 MPa and 3.171 for  $\Delta P = 0.01$  MPa, 8.570 MPa and 3.422 for  $\Delta P = 0.05$  MPa, and 8.542 MPa and 3.55 for  $\Delta P = 0.1$  MPa while the conventional cycle gives an optimum pressure of 8.755 MPa and a COP of 2.82. The entrainment ratio  $w$  increases brutally with gas cooler pressure until it reaches the optimum pressure, after which it increases slightly. However, the entrainment ratio  $w$  is basically independent of the pressure drop.

The effect of the gas cooler outlet temperature on F and CR of the ejector transcritical CO<sub>2</sub> cycle are shown in Fig. 8. It can be seen that the comparison factor F is almost constant while increasing the gas cooler outlet temperature, so the ejector cycle and the conventional cycle performances decrease proportionally to the increase of the gas cooler outlet temperature. The compression ratio CR increases due to the increase of the optimum gas cooler pressure with  $T_{gc}$ . The pressure drop enhances the comparison factor F and increases the compression ratio CR because the optimum pressure drop is above 0.1 MPa. The entrainment ratio  $w$  decreases linearly with  $T_{gc}$  from 0.56 at 33°C to 0.39 at 60°C.

The effect of the evaporation temperature on F and CR of the ejector transcritical CO<sub>2</sub> cycle are shown in Fig. 9. It can be seen that the comparison factor is almost constant while decreasing the evaporation temperature, so the ejector cycle and the conventional cycle performances decrease proportionally with the evaporation temperature reduction. The optimum pressure and the entrainment ratio vary slightly, respectively, around  $8.6 \pm 0.2$  MPa and  $0.54 \pm 0.02$  for pressure drops between 0.01 and 0.1 MPa and evaporation temperature between -5 and 15°C. The CR decreases with the increase of evaporation temperature. With lower evaporation temperatures, the evaporating pressure decrease, so the ejector expansion process increase because the gas cooler optimum pressure is almost constant, yielding to a compression ratio increase. F and CR increase with pressure drop because the optimum pressure drop is above 0.1 MPa.

The effect of the evaporator outlet superheat on F and CR of ejector transcritical cycle are shown in Fig. 10. It can be seen that an increase in the superheat increases F because the superheat penalizes the performance of the conventional cycle more than the ejector one. In the conventional cycle, the compressor suction is superheated which increases the entropy at the suction port and the compression work, while in the ejector cycle, the suction is in saturated vapor and the secondary stream is superheated, which penalizes the entrainment ratio  $w$  because of the mixture quality increase. Increasing the evaporator superheat will decrease the specific evaporator capacity for both the conventional cycle and the ejector cycle. The compression ratio is almost constant, increasing slightly, with the superheat. The optimum gas cooler pressure is almost constant around 8.6 MPa for different pressure drops of the ejector. The comparison factor and the compression ratio increase with pressure drop because the optimum pressure drop is above 0.1 MPa.

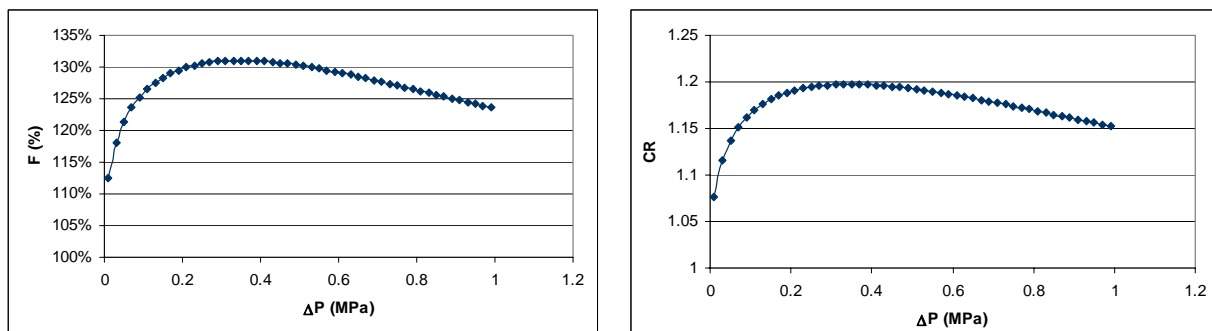


Fig. 6: Comparison factor F and optimum gas cooler pressure versus pressure drop of the ejector.  
 $T_S = 5K$ ,  $T_{ev} = 2^\circ C$ ,  $T_{gc} = 35^\circ C$ .

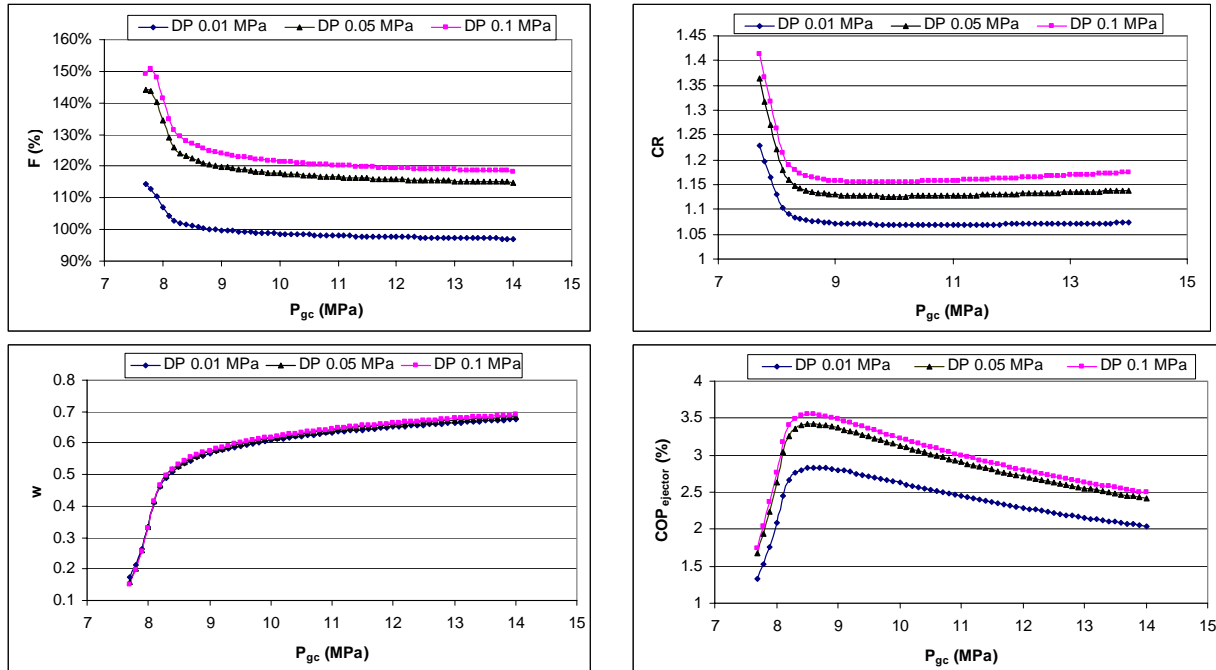


Fig. 7: COP,  $w$ , Comparison factor  $F$  and compression ratio  $CR$  versus gas cooler pressure.  
 $TS = 5K$ ,  $T_{ev} = 2^\circ C$ ,  $T_{gc} = 35^\circ C$

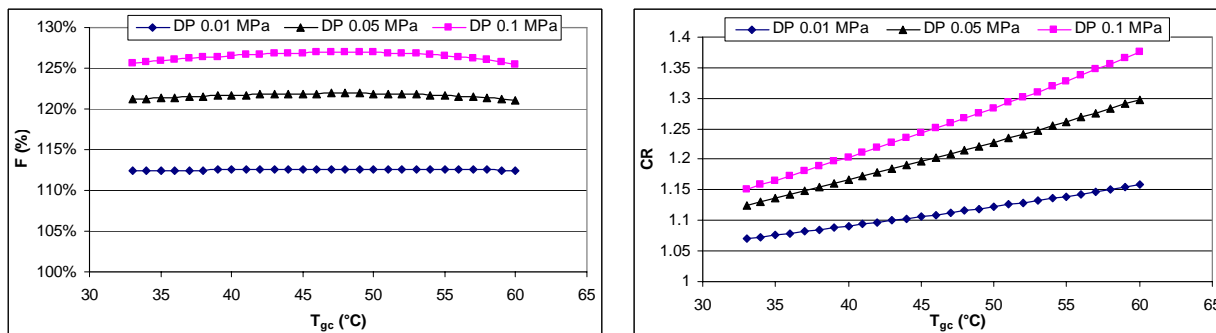


Fig. 8: Comparison factor  $F$  and compression ratio  $CR$  versus gas cooler outlet temperature.  
 $TS = 5K$ ,  $T_{ev} = 2^\circ C$ .

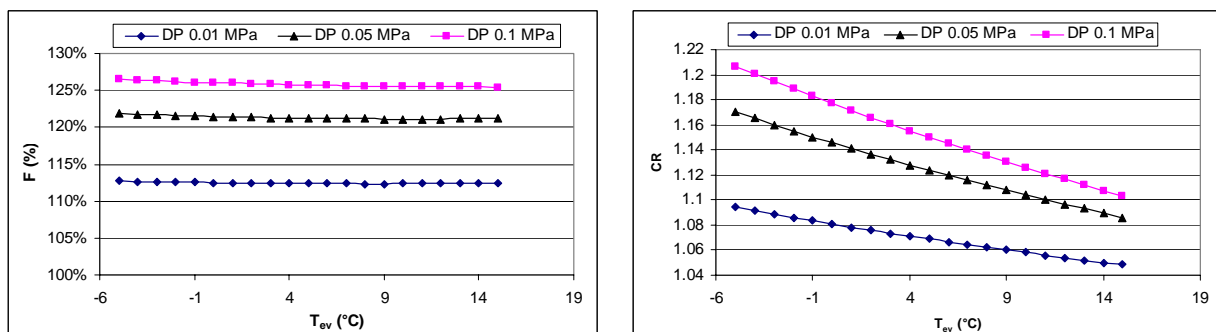


Fig. 9: Comparison factor  $F$  and compression ratio  $CR$  versus evaporator temperature.  
 $TS = 5K$ ,  $T_{gc} = 35^\circ C$ .

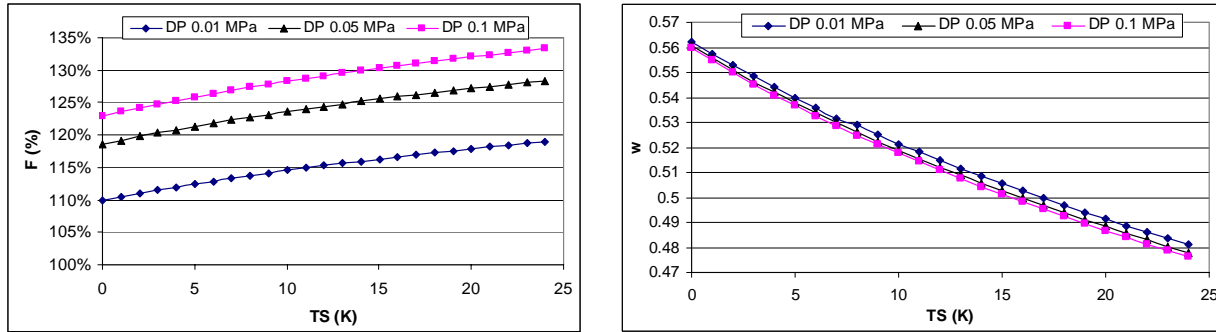


Fig. 10: Comparison factor  $F$  and compression ratio  $CR$  versus evaporator outlet superheat.  
 $T_{gc} = 35^{\circ}\text{C}$ ,  $T_{ev} = 2^{\circ}\text{C}$ .

## 5. PERFORMANCE COMPARISON OF FLUIDS: $\text{CO}_2$ AND R-134a

When the ejector expansion cycle works in sub-critical conditions (condenser outlet in liquid phase), the cycle performances are degraded in the  $\text{CO}_2$  case. Tables 2 and 3 show the performance of  $\text{CO}_2$  and R-134a in conventional cycles and ejector expansion cycles.

The following standard operating conditions are assumed:  $T_{ev} = 2$  and  $-10^{\circ}\text{C}$ ,  $TS = 5$  K. The ejector efficiencies are:  $\eta_n = \eta_s = 0.85$ ,  $\eta_d = 0.75$ . The isentropic compression efficiency is 0.8 for  $\text{CO}_2$  and 0.7 for R-134a.

Table 2: Optimum operation of  $\text{CO}_2$  ejector cycle

$\text{CO}_2$ $\eta_{\text{comp}} = 0.8$ $T_{\text{evap}} = 2^{\circ}\text{C}$										
$T_{\text{evap}}^{\circ}\text{C}$	$T_{gc} / \text{condenser}^{\circ}\text{C}$	$\text{COP}_{\text{ejector}}$	$\text{COP}_{\text{conv}}$	$P_{gc \text{ ejector}} \text{ MPa}$	$DP \text{ MPa}$	$P_{\text{mixture}} \text{ MPa}$	$P_{\text{out ejector}} \text{ MPa}$	$w$	$P_{gc \text{ conv}} \text{ MPa}$	$F (\%)$
2	22	8.182	6.856	6.0	0.1440	3.728	3.922	0.72512	6.0	119.3%
2	28	5.611	4.430	6.892	0.2290	3.7954	4.1171	0.61420	6.892	126.7%
2	30	4.833	3.684	7.214	0.2708	3.8484	4.2405	0.55002	7.214	131.2%
2	35	3.696	2.820	8.513	0.3467	3.917	4.399	0.53528	8.755	131.0%
-10	35	2.653	1.995	8.608	0.3763	2.9476	3.4182	0.51332	8.968	133.0%
2	45	2.451	1.807	11.136	0.5477	4.1732	4.8502	0.46823	11.682	135.7%

Table 3: Optimum operation of R134a ejector cycle

R-134a $\eta_{\text{comp}} = 0.7$ $T_{\text{evap}} = 2^{\circ}\text{C}$										
$T_{\text{evap}}^{\circ}\text{C}$	$T_{gc} / \text{condenser}^{\circ}\text{C}$	$\text{COP}_{\text{ejector}}$	$\text{COP}_{\text{conv}}$	$P_{gc \text{ ejector}} \text{ MPa}$	$DP \text{ MPa}$	$P_{\text{mixture}} \text{ MPa}$	$P_{\text{out ejector}} \text{ MPa}$	$w$	$P_{gc \text{ conv}} \text{ MPa}$	$F (\%)$
2	22	8.949	8.492	0.608	0.0060	0.316	0.324	0.84786	0.608	105.4%
2	28	6.771	6.322	0.727	0.0096	0.3175	0.3302	0.80964	0.727	107.1%
2	35	5.220	4.773	0.887	0.0149	0.320	0.340	0.76517	0.887	109.4%
-10	35	3.568	3.169	0.887	0.0179	0.2080	0.2306	0.70958	0.887	112.6%
2	45	3.862	3.409	1.160	0.0247	0.3259	0.3577	0.70148	1.160	113.3%

For  $\text{CO}_2$  at  $2^{\circ}\text{C}$ , the optimum ejector cycle performances decrease when the gas cooler temperature increases; the comparison factor  $F$ , the optimum pressure drop, the mixture pressure and the ejector outlet pressure increase but the entrainment ratio  $w$  decreases with the increase of the gas cooler pressure, because of the vapor quality increase. That explains the decrease of the ejector cycle.

For R-134a at  $2^{\circ}\text{C}$ , the same behavior appears, but the energy performances are lower than that of  $\text{CO}_2$  ejector cycle, also the pressure drop is smaller due to the thermodynamic properties of R-134a to the pressure.



## 6. CONCLUSIONS

An ejector expansion cycle enhances the performance of a conventional CO<sub>2</sub> cycle in a transcritical process for a gas cooler temperature above 31°C. A constant pressure-mixing-zone model for the ejector has been used to analyze the ejector thermodynamic cycle analysis. The influence of pressure drop at the ejector nozzle and gas cooler pressure on the performances of the ejector cycle is discussed using the model. The performance analysis of ejector cycle versus conventional cycle is performed considering the effect of variation of gas cooler pressure, evaporating temperature, gas cooler outlet temperature and evaporator outlet superheat. The CO<sub>2</sub> ejector cycle improves the COP more than 30% for the usual operating conditions, while the ejector cycle shows lower efficiencies in sub-critical cycles. A COP performance comparison between CO<sub>2</sub> and R-134a has also been done.

## NOMENCLATURE

COP	coefficient of performance	adim.	<i>Subscripts and superscripts</i>	
CR	ejector compression ratio	adim.		
CV	control volume		comp	compressor
F	comparison factor	%	cond	condenser
h	enthalpy	kJ/kg	conv	conventional
L	length	m	d	diffuser
m	mass	kg	e, ev, evap	evaporator
P	pressure	MPa	gc	gas cooler
T	temperature	°C	is	isentropic
TS	superheat	K	L	low
w	entrainment ratio	adim.	m	mixture
<i>Greeks</i>			n	nozzle
Δ	difference		p	primary
η	efficiency	adim.	s	secondary
ρ	density	kg/m <sup>3</sup>	sat	saturation

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